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Theoretical and Experimental Analysis of Torsional and Bending Effect on Four Cylinders Engine Crankshafts by Using Finite Element Approach

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Abstract:- *The problem of torsional vibration of the crankshaft of high-speed diesel engine has become critical with increase in excitation forces. This results in high torsional vibration amplitudes and hence high stresses the paper aims at complete FEM analysis of a crankshaft for torsional and bending vibrations, identification of stresses. It is analyzed for natural frequency, rigid body mode shape by ANSYS and Holzer method. The complete simulation of actual boundary conditions is done for journal bearing support, inertia lumping for reciprocating parts and bearing stiffness. Customized code is developed in ANSYS-Macros, which will convert user input Pressure-Crank angle variation to excitation forces for various orders through FFT. The dynamic responses obtained for displacement and stresses. Finally all results are combined to obtain the variation of Fillet Stress as a function of engine speed and harmonic orders. The critical dynamic response is compared with results obtained experimentally for torsional amplitudes.*

Key Words: -Torsional vibration, Bending vibration, FFT setup, Modal analysis, Rigid body modes, Static analysis, Critical fillet stresses, ANSYS Macro, Material optimization.

Introduction: Till recently crankshaft torsional vibration analysis was done by the empirical formulae and iterative procedures, but the simplifying assumption that a throw of crankshaft has one degree of freedom is only partially true for torsional modes of vibrations. More degrees of freedom are required to get information about other modes of vibration and stress distribution. Since last decade advent of powerful finite element analysis (FEA) packages have proven good tool to accurately analyze them. The complicated geometry of crankshaft and the complex torque applied by cylinders make their analysis difficult. But optimized meshing and accurate simulation of boundary conditions along with ability to apply complex torque provided by various FEM packages have helped the designer to carry torsional vibration analysis with the investigation of critical stresses.FEM enables to find critical locations and quantitative analysis of the stress distribution and deformed shapes under loads. The specific engine crankshaft of a major automobile company (the name is kept confidential) is taken as the model for the analysis. It is a 4-cylinder, 4-stroke, turbocharged diesel engine with in-line throw crankshaft.

Objectives: The project aims at detail FEM analysis of crankshaft. The following are the main objectives of the project.

1. Building a 3-D Solid parametric model of crankshaft, Flywheel and pulley in Pro-Engineer wild fire.

2. Meshing the model by Tetrahedral Solid 45 elements in Hypermesh. MPC 184 element is also used to define the journal bearings.

3. Rigid body modes and Normal modes- we have calculated in free vibration analysis for Crankshaft, Crankshaft + Pulley, Crankshaft + Flywheel, Crankshaft + Pulley + Flywheel.

4. Experimental modal analysis carried out using FFT ANALYSER MACHINE to validate our CAE work.

5. Behaviour of torsional modes, bending modes and combined modes of vibration we have studied for all systems.

6. ANSYS Macro is used to find out the Ft and Fr forces from given P and Ɵ diagram.

7. C programming is used to find out the T and Ɵ diagram from corresponding P and Ɵ diagram.

8. Dynamic response and investigation of critical stresses are found out by incorporating the boundary conditions at the journal bearing positions and tangential and radial forces on corresponding nodes.

9. Refinement and restudy of the stresses at the fillet regions done by refining a pair of crank lobes and journal bearings.

For this HYPERMESH used for meshing and ANSYS used as solver.

Crankshaft Vibrations: In I.C. engines various types of excitation forces exist. These directly or indirectly affect the crankshaft dynamics. [1]

The major types of these vibrations are

• **Torsional vibrations:** In multi-cylinder engine crankshafts, the crank throws are spatial or out of phase with each other for the balancing purpose. It is also attached with a flywheel and some driven system. The torque is applied to the crankpin by the connecting rod. This torque is of varying nature because of variation in gas pressure and inertia forces. The fluctuating torque at the crankpin causes the twisting and untwisting periodically. Hence the torsional vibrations are induced.

• **Flexural vibrations:** The lateral periodic motion of crankshaft under the fluctuating forces exerted by connecting rod at crankpin cause bending vibrations of crankshaft. This mode shape generally has many nodes because the bending vibrations are strongly reacted at the bearings.

• Axial vibrations: The torsional vibrations can cause axial vibration in the twisting and untwisting motion. Also radial forces at crankpin cause some axial movement of crank throw.

• Coupled vibrations: In general, however the various modes of vibration are coupled so that vibrations of one type can't occur without an accompanying vibration of the other type. These are not troublesome if there is considerable spread between the natural frequencies of the modes of vibration involved; i.e. the modes get weakly coupled.

Influence of Crankshaft Vibrations:
The crankshaft vibrations badly affect the working of engine. The major areas are as follows.

1. The torsional vibrations cause the angular velocities of all the cranks to vary but not in the same proportions. The crank away from the node has maximum effects compared to crank near the node. This affects the balancing.

2. Due to same reason discussed above, stresses of varying intensity are generated in whole length of the crankshaft. These are also fluctuating in nature and hence cause fatigue of crankshaft, reducing its life. The stresses induced are dangerous at fillet or oil-hole locations.

3. Vibratory energy is transmitted to all parts of the structure where it causes structural damage.

4. It induces noisy operation of engine, which is undesirable in passenger cars. It also causes wear of all running parts. [2]

Modal Analysis By Holzer Method:
This analysis, though not accurate, greatly helps in predicting the torsional vibration behaviour of the system. The reference [1] had developed many empirical formulae to model the crankshaft as a lumped system consisting of shafts and disks. Corresponding stiffness, inertias and the equivalent length calculations can be done using these formulae.

Results:

The natural frequencies for above system in Hz are,Table 1. Comparison of results.

Mode. No.	**7**	**8**	**9**	**10**	**11**
Natural Frequency by Holzer method	**433.2**	**902.42**	**993.44**	**1049.34**	**1132.12**
Natural Frequency by ANSYS	**408.87**	**571.47**	**937.11**	**1047.43**	**1054.34**

From above table we found the ANSYS results are agree with Holzer method results.

Solid Modeling of Crankshaft:
As a prerequisite to the finite element model is the physical geometry of the part i.e. the suspension link we have created using Prove-Wildfire software.

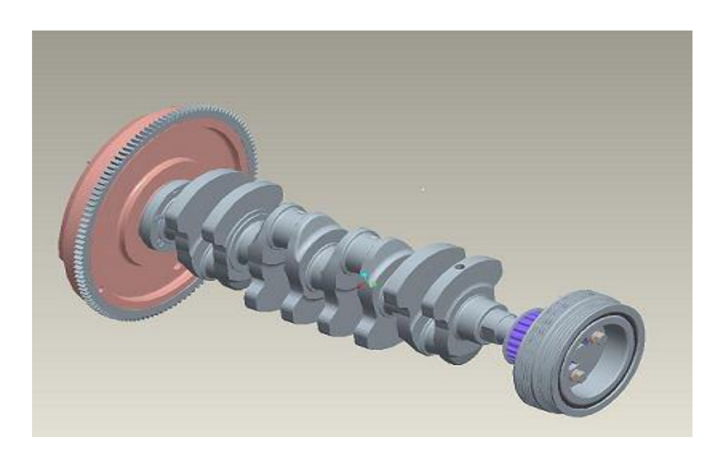


Fig.1. Model of whole assembly

FE Mesh Generation:
Meshing generally falls in two categories depending on the geometry of the element. For a 3D machine element of regular shape, solid meshing is sufficient, but for irregular geometries we have to first use surface meshing and then solid meshing. For surface meshing the solid model is imported in HYPERMESH (software exclusive for meshing).We have done surface meshing first by Plane-42 element. One of the important aspects in surface meshing is merging of nodes or technically it can be called node equivalence. Once the node equivalence is confirmed we have to go for free-edge checking which will ensure that there are no free surfaces.

Now, after free surface check is done, we go for quality check, the quality check in HYPERMESH is a unique feature as it allows us not only to check for internal and external

angles of the mesh element but also facilitates in checking aspect ratio, warpage ratio, skew ratio, and most important the Jacobian matrix. The value of the Jacobian matrix should always lie between 0 and 1. Any other value of the Jacobian matrix renders the element faulty and a new element should be created by deleting the previous one. In our case the value of Jacobian matrix is 0.7 once assured with a safe and sound surface meshing our next step is to import the model in ANSYS for solid meshing.

The element used for solid meshing is 10 Node Solid 92 Tetrahedral Element. The special features of this element are Plasticity, Creep, Swelling, Stress stiffening, large deflection, large strain, Birth and death. [4]

MPC184

Multipoint Constraint Elements: Rigid Link, Rigid Beam, Slider, Spherical, Revolute, Universal MPC184 comprises a general class of multipoint constraint elements that implement kinematic constraints using Lagrange multipliers. The elements are loosely classified here as “constraint elements” and “joint elements”. All of these elements are used in situations that require you to impose some kind of constraint to meet certain requirements. The constraint may be as simple as that of identical displacements at a joint. They can also be more complicated, such as those that involve rigid modeling of parts, or kinematic constraints that transmit motion between flexible bodies in a particular way. For example, a structure may consist of some rigid parts and some moving parts connected together by some rotational or sliding connections. The rigid part of the structure may be modeled using the MPC184 Link/Beam elements, while the moving parts may be connected with the MPC184 slider, spherical, revolute, or universal joint element. Since these elements are implemented using Lagrange multipliers, the constraint forces and moments are available for output purposes. This element is used to define the connectivity between the point of application of force and the nodded on the surface of the structural member.

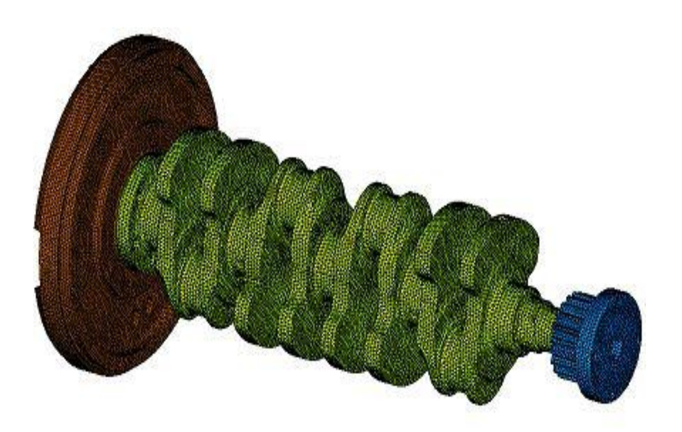


Fig.2. Meshed model of whole assembly generated in Hypermesh

Fig. 3. Enlarge view of dense meshed model

Experimental Setup for Natural Frequency:
The FFT analyzer used is dual channel FFT (Fig.4) analyzer 2900 B from Larson & Davis Company Ltd. It is full function, yet portable, time/frequency signal analyzer. In this experimental analysis, we hang the crankshaft at its C.G. and excitation is given by hammer. Then we get results from FFT, which are given in table 2.



Fig. 4. Experimental setup

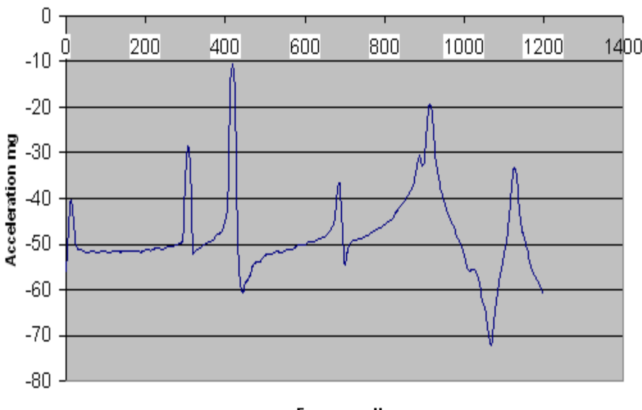


Fig. 5 Frequency Vs Magnitude by Normal Function

Table 2. Comparison for Crankshaft and Pulley

Mode	ANSYS Results	FFT Results	% Error
7	337.4001	306.25	7 %
8	452.6897	418.75	7.5 %
9	757.2222	687.50	8 %
10	972.9449	918.75	4 %
11	998.1289	1100.00	9 %

From above table we see that Ansys results are agree with the experimental results. So our software results are accurate and these results are used for further analysis.

Torsion and Bending Mode Analysis:

The effect of different assembled components on the crankshaft mode shape and natural frequency is our objective.

System 1: A simple crankshaft (free-free) without flywheel, pulley and bearings.

System 2: System 1 is modified by including the flywheel.

System 3: System 1 is modified by including the pulley.

System 4: System 3 is modified by incorporating the flywheel and pulley As we increase the mass of system by adding extra pulley and flywheel the decrease in frequency in each mode and the deformation plot is given for a better comparison of the effect on bending, torsion and combined mode due to addition of flywheel at one end.

Analysis for Crankshaft, Pulley and Flywheel:

Table 3. Natural frequencies.

7	222.3451359827	First Bending Mode
18	1536.908870673	First Torsion Mode
19	1664.162853562	First Combined Mode

A) Deformation Plot For 1st Bending Mode of

Crankshaft, Flywheel & Pulley System

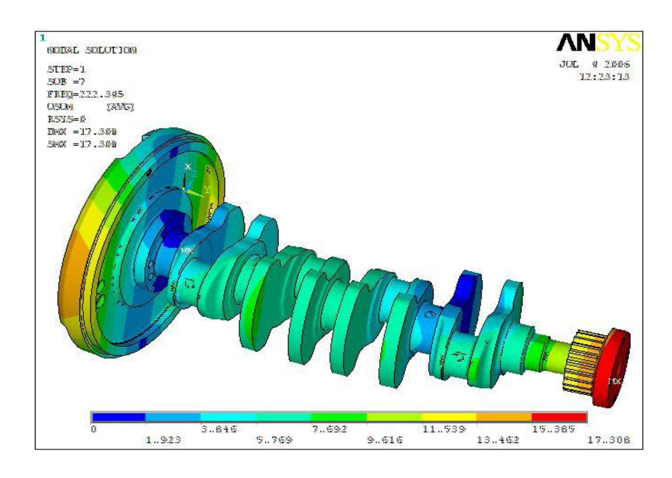


Fig. 6 Deformation plot

From above plot we found the maximum deflection is 17.308 mm and minimum deflection is 0 mm.

B) Deformation Plot for 1st Torsion Mode of Crank Shaft, Flywheel & Pulley System.

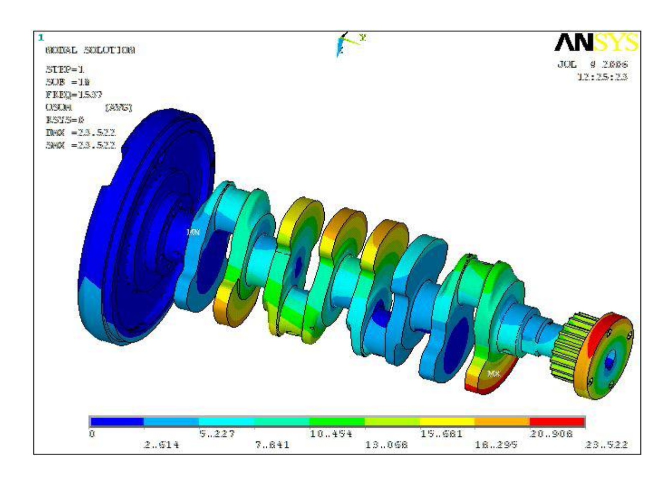


Fig. 7 Deformation plot

From above plot we found the maximum deflection is 23.622 mm and minimum deflection is 0 mm.

C) Deformation Plot for 1st Combined Mode of Crank Shaft, Flywheel and Pulley System.

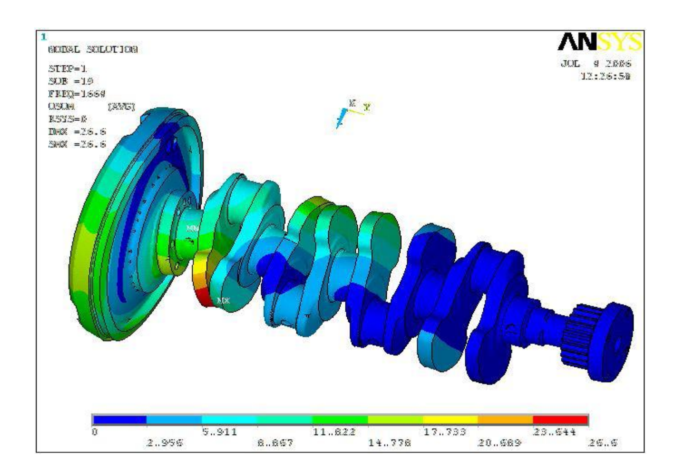


Fig. 8 Deformation plot

From above plot we found the maximum deflection is 26.6 mm and minimum deflection is 0 mm.

Force Application:
We get data about crank angle (Θ) and corresponding pressure from company. Further we calculate the data of crank angle (Θ) and corresponding torque using 'C' programming language.

- Output of C program:

Figure 8.3 shows output of C program.

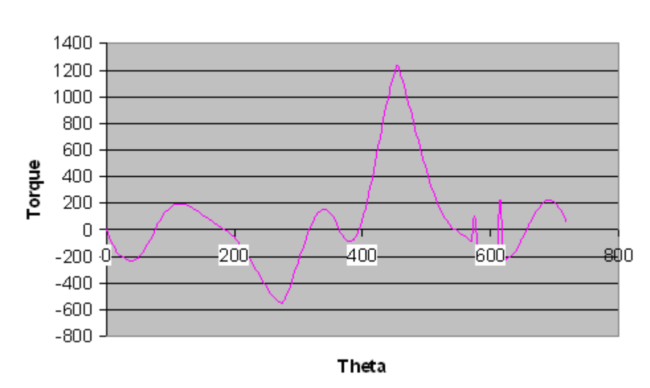


Fig. 9 Torque Vs Theta graph.

ANSYS MACRO FOR CALCULATING TANGENTIAL FORCE (Ft) & RADIAL FORCE (Fr):
We can record a frequently used sequence of ANSYS commands in a macro file (these are sometimes called command files). Creating a macro enables you to, in effect; create your own custom ANSYS command.

Creating a Macro:

You can create macros either within ANSYS itself or using your text editor of choice (such as emacs, vi, or WordPad). If your macro is fairly simple and short, creating it in ANSYS can be very convenient. If you are creating a longer, more complex macro or editing an existing macro then you will need a text editor. Also, using a text editor allows you to use a similar macro or ANSYS log file as the source for your macro.

Multi-Prompt for Variables
Engine data - Cranktrain Parameters
Engine Stroke (m)
Stroke 0.09
Engine Bore Diameter (m)
EBore_di 0.08312
Connecting Rod Length (m)
Crodleng 0.145
Engine Speed (rpm)
ESpeed 4300
OK Cancel

Fig. 10 Input windows for crank pin parameters

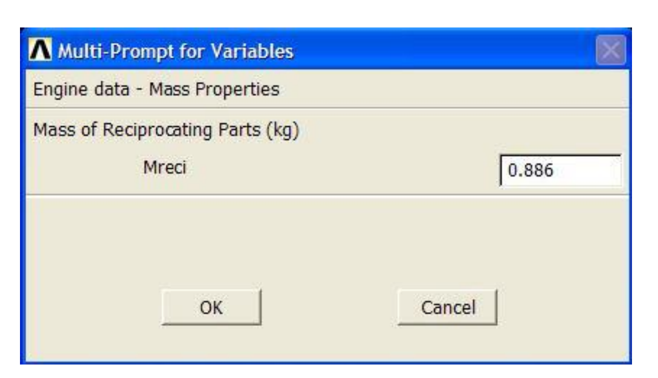


Fig. 11 Input windows for mass properties of reciprocating parts

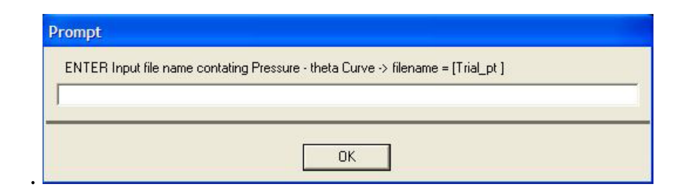


Fig. 12 Input windows prompt for p-o values

Output Table for Ft, Fr Torque From Macros:
Crank Radius (m) = 0.0450

Connecting Rod Length (m) = 0.1450

Engine Bore Diameter (m) = 0.0831

Piston Area (m^2) = 0.0054

Engine Speed (rpm) = 4300.00

Angular Speed (rad/s) = 450.4762

Mass of Reciprocating Parts (kg) = 0.8860

Table 4 some of the values of output



Crank Angle (Deg.)	Tan. Force (F_t) (N)	Radial Force (F_r) (N)	Output Torque (N-m)
0	0.00000	-10601.700	0.000
5	-1197.918	-10428.895	-53.906
10	-2325.359	-9997.010	-104.641
15	-3324.537	-9328.659	-149.604
20	-4145.564	-8460.840	-186.550
25	-4748.502	-7439.494	-213.682

Critical Fillet Stress Studies:

To predict dynamic behaviour of any system, the systems around the model have to be considered and simulated mathematically. It contains forces, constraints, reactions applied by external system on the system under consideration. In the terminology of FEM it is called 'Boundary Conditions', on the basis of which the model is solved.

Analysis for Radial Force:

We get radial and tangential force values from

ANSYS macro. Further we apply these forces on crankshaft and solve it with ANSYS solver. To get the effect of bending forces on the crankshaft, we have applied the radial forces Fr on the crankpin positions and the displacement constraints is applied at the multiple journal bearing positions. The study of maximum stress distributions around the fillet, which may be stress concentration region, is our objective. Figure 13 shows application of radial force.

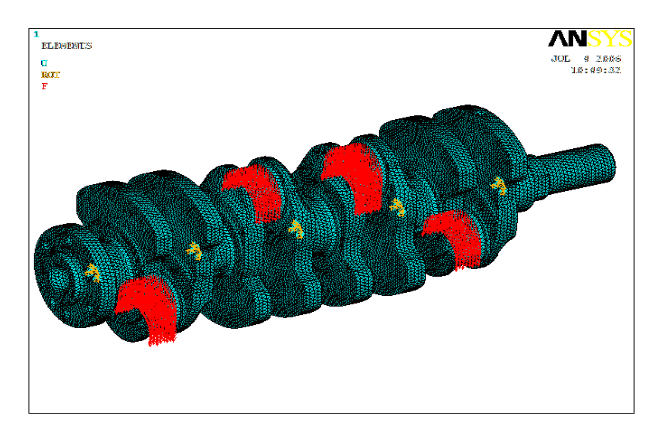


Fig. 13 Boundary Conditions for Radial Stress

Analysis.

- Plot For Stress Due To Radial Forces:

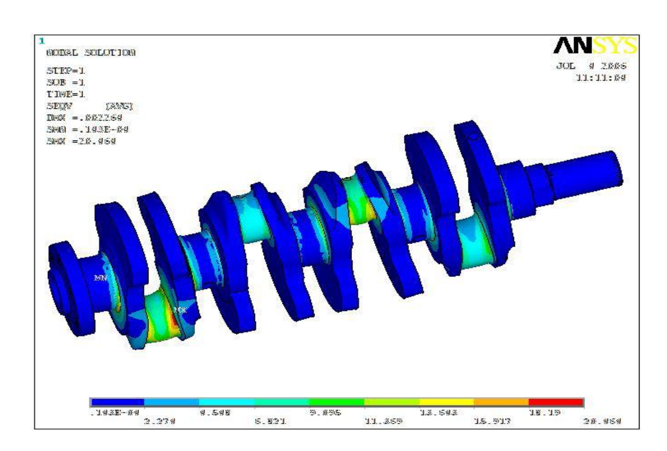


Fig. 14 Von mises stress plot

From above plot we found the maximum von mises stress value is 20.464 N/mm2 and minimum stress value is 0.19E-4 N/mm2.

Analysis for Tangential Force:

We get tangential force values from ANSYS macro, and further we apply these forces on crankshaft. To get the effect of torsional forces on the crankshaft, we have applied the tangential forces Ft on the crankpin positions and the displacement constraints is applied at the end of one side journal bearing positions to see the effect of toque developed by tangential forces at different lobes of crankshaft. The study of maximum stress distributions around the crankpin positions, which may be stress concentration region, is our objective. Following figure 15 shows tangential force application on crankshaft.

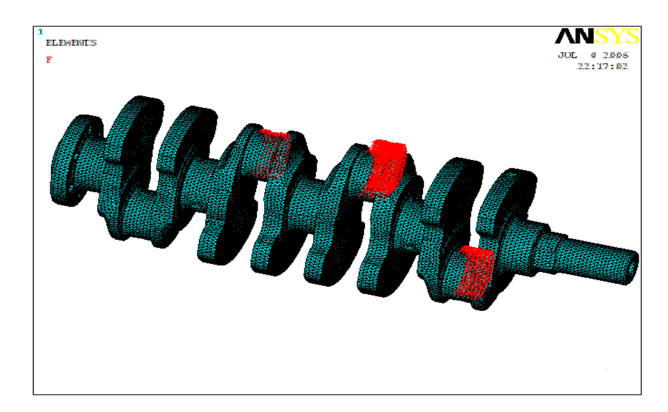


Fig.15 Application of tangential force.

- Plot for Stress Due To Tangential Forces:

Fig. 16 Von mises stress plot from above plot we found the maximum von mises stress value is 20.464 N/mm2 and minimum stress value is 0.19E-4 N/mm2

Material Optimization:

To study the effect of radial and tangential forces on different material conditions for same geometrical parameters of crankshaft and to study stress distribution around the stress concentration area like fillet stresses is our objective for material optimization study.

1) For 40 Cr 4 MO3:

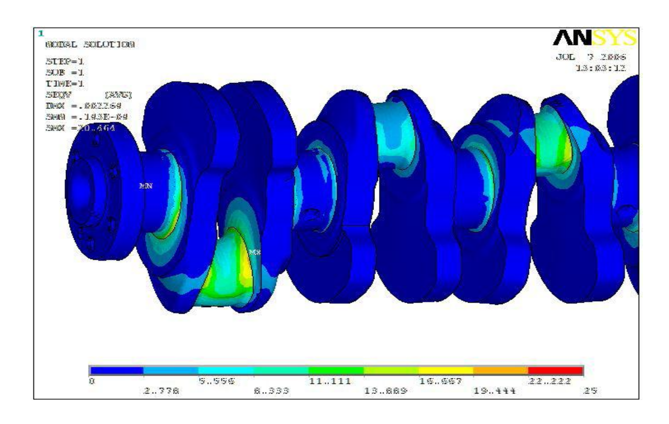


Fig. 17 Von mises stress plot

2) MED Carbon, B, Mn, V, Alloy:

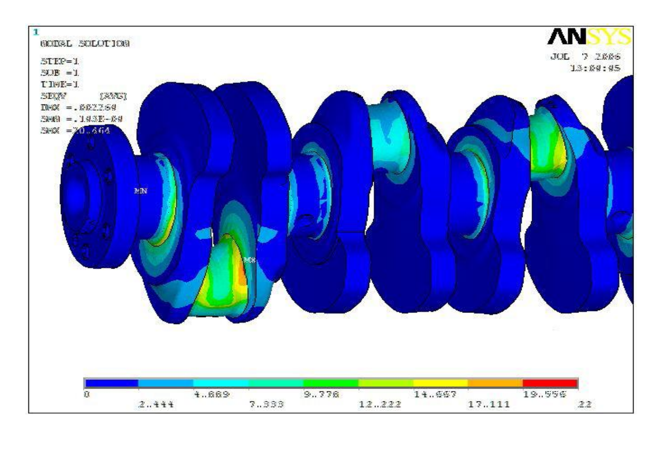


Fig. 18 Von mises stress plot

Discussion and Conclusion

1. Experimental validation of crankshaft is done using FFT analyser. The values of FFT output and Modal frequency calculated from ANSYS and Hypermesh validates the results as both are matching.

2. The mode shape calculation of different systems of component is done using modal analysis. The effect of flywheel and pulley shows the natural frequency is decreasing due to additional masses at the end of crankshaft.

3. The bending mode, torsion mode and combined mode frequency values calculated by Ansys shows the actual working condition deformation during high speed rotation rpm value of crankshaft.

4. As the stress concentration area for radial forces is fillet and for tangential force e it is crankpin positions. Both the stresses plot study shows the critical stress region.

5. As the crankshaft is already optimized at the fillet area at the end, keeping all geometrical parameter of crankshaft same, we studied the maximum stress area for the different materials used for four to five currently vehicles on road.

6. We conclude our results and discussion by viewing displacement plot for different mode shapes that the natural frequency is affected or decreased due to the addition of pulley and flywheel. So the resonance condition is more critical for the combined assembly. The Vonmises stress plot concludes that during maximum rpm or maximum torque condition failure will take place near the fillet area.

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